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Biofuel and Gas Turbine Engines

Marco Antônio Rosa do Nascimento and Eraldo Cruz dos Santos
Federal University of Itajubá – UNIFEI
Brazil

1. Introduction

Currently, the interest in using vegetable oils and their derivatives as fuel in primary drives for the generation of electricity has increased due to rising oil prices and concerns over the environmental impacts caused by fossil fuel use. For viability of using biodiesel as a substitute for fossil fuels for power generation, should be considered the emissions of greenhouse gases, i.e., pollutants such as nitrogen oxides (NO_X), sulfur oxides (SO_X), carbon monoxide (CO) and particulates into the atmosphere during the lifetime of the power plant.

According HABIB (2010) the effect of using petroleum-derived fuel in aviation on the environment is significant. Given the intensity of air traffic and civil and military operations, making the development of alternative fuels for the aviation sector is justified, necessary and critical.

Another concern that must be considered is the quality of biofuel to be stored over time, this being an obstacle to be overcome in order to maintain fuel quality and operational reliability of gas turbine installations operating with biofuels.

Biofuels also have the advantage of being renewable and cleaner, this is due in large part because they do not contain sulfur in its composition. The use of distributed generation renewable fuels can be advantageous in isolated regions, far from major urban centers, to generate electricity using the resources available on site.

Among other engines, gas turbines represent one of the technologies of distributed generation, which is characterized by the supply of electricity and heat simultaneously. In principle these machines should operate without major problems by using biofuels, because of similarities with the characteristics of the fuels conventionally used. However, there are few references on the performance of gas turbines operating on biofuels and this is the motivation of this study.

Microturbines are small gas-turbo generators designed to operate in the power range from 10 to 350 kW. Although its operation will also be based on the Brayton cycle, they present their own characteristics that differentiate them from large turbines.

Most gas turbine available today, originated in the military and aerospace industry. Many projects were aimed at applications in the automotive sector in the period between 1950 and 1970. The first gas turbine generation was developed from turbo aircraft, buses and other commercial means of transport (SCOTT, 2000). Interest in stationary generation market has expanded in the years 1980 and 1990, and its use in distributed generation has been accelerated (LISS, 1999).

It is hoped that in future gas turbines of small power is an alternative for power generation for residential and commercial segment, since the operational reliability is one of the main needs in these sectors (WILLIS and SCOTT, 2000). These turbines have various applications such as power generation in the place of consumption (on-site), the uninterrupted supply of electricity, to cover peak loads, cogeneration and mechanical drive, which characterizes the distributed generation (BIASI, 1998).
Gas turbines may use different types of fuel such as diesel, kerosene, ethanol, natural gas and gas obtained from biomass gasification, etc. The shift to gas from biomass has been considered promising, but some changes must be made in supply and combustion systems turbine, aiming to modify the injection and control systems and the volume of the combustion chamber.
The scope of this chapter includes a brief description of the systems of gas turbines, reports the experiences of biofuel use in gas turbines made until today, with emphasis on the experience developed in the Laboratory of Gas Turbines and Gasification the Institute of Mechanical Engineering, Federal University of Itajubá - IEM/UNIFEI aspects of thermal performance and emissions of gases from gas turbines of small power.

2. Biofuels

Biofuels are fuels of biological origin, i.e., not fossil. They are produced from plants such as corn, soy, sugar cane, castor beans, sugar beet, palm oil, canola, babassu oil, hemp, among others. Organic waste can also be used for the production of biofuel. The main biofuels are ethanol (produced from sugar cane and corn), biogas (biomass), bioethanol, biodiesel (from palm oil or soy), among others.
Biofuels can be used on vehicles (cars, trucks, tractors, etc.), turbines, boilers, etc.., in whole or blended with fossil fuels. In Brazil, for example, soy biodiesel is blended with fossil diesel. Is also added to gasoline the ethanol produced from sugar cane.
The advantage of using biofuels is the significant reduction of greenhouse gas emissions. It is also advantageous because it is a renewable source of energy instead of fossil fuels (diesel, gasoline, kerosene, coal).
This section will describe some characteristics and requirements of biofuels that have potential for use in gas turbines.

2.1 Gas turbines operating on liquid fuels

Biofuels have the greatest potential for use in gas turbines are biodiesel and ethanol, due to factors such as availability physical-chemical characteristics similar to fossil fuels such as diesel or jet fuel. Table 1 presents a summary of requirement of liquid fuel as defined by the manufacturers of gas turbines for efficient operations (BOYCE, 2006).

Moisture and sediment	1.0 % (v%) maximum
Viscosity	20 cS at injector
Dew-Point	20 °C at ambient temperature
Carbon Residue	1.0 % (p.) maximum
Hydrogen	11% (p.) maximum
Sulfur	1% (p.) maximum

Table 1. Requirements liquid fuel for gas turbines.

The growing interest in biofuels along with increasing market demand for generators supplied by renewable fuels has led manufacturers to modify the designs of gas turbines and micro-turbines, in order that they can operate on biofuels.

For biodiesel, the supply system is being modified to fit this new biofuel due to some reasons such as higher viscosity, content of acylglycerols and the effects of corrosion. New corrosion-resistant materials, systems control the flow of fuel and improved geometry optimized for the guns are some of the challenges of these new projects.

In scientific literature there is little information about testing of gas turbines for small power, operating on biofuels. To study the impact of biofuel use in the operation and maintenance of gas turbine, one must take the following measures:

- Define the physical and chemical characteristics of both diesel and biofuel used in the tests. Some important characteristics are: density, distillation, viscosity, ash content, phosphorus, iodine and sulfur, water content, cetane number, oxidation stability, flashpoint, freezing point, dew point, volumetric composition of methyl, ethyl and lipids, glycerol, lower calorific value, etc. These values should be compared with the requirements of the standards on diesel and biodiesel to demonstrate that they can be used in the study.
- Once further tests it is possible to define what characteristics of biodiesel are relevant to the determination of changes in engine behavior, considering the performance parameters and emissions. The higher viscosity of biodiesel can lead to difficulties in its injection into the combustion chamber. It is possible reduce the viscosity of the mixture increases its temperature, or by adding alcohol. The lower the flash point of biodiesel could also cause problems in combustion.
- It is possible find accumulation of carbonized material in the inner parts of the gas turbine, after the tests with biodiesel. The biofuel can produce corrosion in fuel supply system.
- It is also recommended to install a filter at least 50 μm at the fuel supply in the gas turbine when using biodiesel.
- It is not advisable to use biofuels in gas turbines without performing a preliminary economic analysis.

Some alternative liquid fuels such as vegetable oil, biodiesel or pyrolysis oil, ethanol and methanol are being tested in gas turbines (GÖKALP, 2004).

As biodiesel has similar properties to diesel, it can be used directly in a gas turbine, blended with diesel in various proportions (usually uses 5 to 30% biodiesel in the blend with diesel). The properties of biodiesel are slightly different to those of diesel in terms of energy content or physical properties.

The Lower Heating Value (LHV) of liquid biofuels such as pure biodiesel (B100), B5 B30 and vegetable oils are between 37,500 and 44,500 kJ/kg, which is close to regular diesel (GÖKALP, 2004).

The viscosities of ethanol, biodiesel and its blends with diesel are lower than the residual oil from the kitchen, making it easier to spray. Vegetable oils and oils derived from pyrolysis have a very high viscosity, which causes problems in its mist inside the combustion chamber of gas turbines. However, these fuels can be heated to reduce its viscosity before being injected into the combustor.

Fuel oil resulting from pyrolysis of wood, vegetable oils and methyl esters has a carbon/hydrogen rate (C/H) higher than that of conventional diesel. As consequence, there may be an accumulation of soot inside the combustion chamber or turbine blades (GÖKALP, 2004). Another factor that changes as a result of this feature is the transfer of heat by radiation from the flame to the flame tube.

2.2 Gas turbines operating on gaseous fuels

Thermal power plants with gas turbines operating with gas from biomass need to present efficient, simple technology, low cost and operational reliability, in order that these plants could become economically competitive with traditional systems of power generation, such as stationary alternative engines.

The potential of gas turbines for this application is great, although the gas must be subjected to cleaning to remove solid impurities and/or gas that can damage some components of some systems of gas turbine.

Gaseous fuels can be obtained by gasification of biomass, which in addition to the gas generates a set of substances, such as tar which is a compound in gaseous form in the fuel gas, which has an appreciable calorific value, although it shows a tar obstacle to the use of gaseous fuels in internal combustion engines, due to its high corrosive power of components and reduced engine efficiency.

In the case of gas turbines, the tar can be a problem only happens when its condensation. Basically there are two strategies to address the problem of tar, remove it from the fuel gas or burn it in the combustion chamber. In the first case, nickel-based catalysts have shown very promising results. In the second case, the strategy is to keep the fuel temperature above the dew point of tar in the gas supply pipes, and perform your burning at high temperature in the combustion chamber (SCHMITZ, 2000).

Currently, gas turbines are designed for a specific fuel (natural gas or fuel oil). Recent progress has been achieved in the methodologies and tools for the design of combustors for gas turbine. It is possible to perform a clean combustion of fossil fuels by employing low-carbon technologies based on premixed combustion.

There are ongoing projects that aim to harness these advances for applications geared to a wider range of fuels with commercial potential, including those with low calorific value, obtained from biomass gasification. Some procedures should be established for selecting appropriate fuels to be used taking into account the performance of combustion and emissions of soot and NO_X. Furthermore, it should be considered the adaptability of existing burners to use alternative fuels selected (GÖKALP, 2004).

Some gaseous alternative fuels also have potential for use in gas turbines, for example, the synthesis gas from gasification of biomass, the biomass pyrolysis gas, gas from digesters (biogas) and residual gas from industrial processes, which are rich in hydrogen.

Industrial gases such as methane reformed with steam, refinery gas, residual gas from the Fischer-Tropsch gasification gas with oxygen gas and slow pyrolysis of wood, have a LHV comparable to natural gas. This is due to the high hydrogen content of fuel gases, which lies between 19 and 45% of the volume. Rather, the LHV of gas gasification with air and biogas are very low because they are produced at atmospheric pressure, so they must be compressed before being used in gas turbines.

Except reformed methane with steam, all other gaseous fuels mentioned above have a C/H greater than that of natural gas (GÖKALP, 2004).

According BOYCE (2006) in Table 2 presents an overview of the requirements for gaseous fuels that can be used in gas turbines.

Calorific	11,184 – 41,000 kJ/m³
Solid Contaminants	< 30 ppm
Flammability Limits	2.2 : 1
Content of sulfur, sodium, potassium and lithium	< 5 ppm (In the form of meta alkaline sulfate)
H_2O (p.)	< 25%

Table 2. Requirements gaseous fuel for gas turbines.

According KEHLHOFER (2009) the type and composition of the fuel has a direct influence on efficiency and emission of gases from a gas turbine. The LHV of the fuel is important because it defines the mass flow of fuel and consequently its specific consumption. The fuel composition is also important as it influences the performance of the cycle because it determines the enthalpy of gas entering the turbine, and the available enthalpy drop off the engine and the amount of steam generated in the recovery boiler.

2.3 Clean fuel gas

Gaseous fuels obtained through the processes of gasification and pyrolysis produce fuels with low-and medium calorific value. According to Table 1, in some gas turbines require a minimum value for the calorific value of gas, which can be difficult to achieve it.

Manufacturers set very strict parameters regarding the quality of the combustible gases to avoid possible damage to the hot parts of gas turbine. Alkaline components present in the fuel, especially chlorides cause corrosion at high temperatures. The presence of tar should also be considered if the operations of fuel valves occur at temperatures below the dew point of the tar.

Almost all biogenic fuels contain considerable amounts of halogens (Cl, F, Br), alkali (Na, K), alkali oxides and other metals (Zn, Cu, Ca). The sulfur content is usually low. Unless most of these items may be retained in the gasifier or pyrolysis reactor, the requirements of the fuel gas cannot be met when employing biomass. The fuel gas contains impurities mentioned in the gaseous or solid, so it is inevitable that they would perform a deep cleaning of the gas. The chloride concentration should be considerably reduced. The slow pyrolysis produces gases, which are characterized by average values of calorific value, situated in the range between 7,000 and 13,000 kJ/kg, and low levels of chlorides and alkalis.

Should be applied under heating rates and moderate temperatures to produce pyrolysis gas with low alkali chloride. High rates of heating fuel break biogenic structure, which favors the conversion of solid pyrolysis gas, however, compromises the retention of impurities in the vegetable coal. High levels of retention in coal are only achieved with low rates of heating in combination with moderate temperatures.

It is estimated that for slow pyrolysis with a maximum temperature of 350 °C are retained approximately 86% of chloride, and the calorific value of gas reaches 10.9 MJ/kg, which can be considered average. These conditions result in a lower ratio chloride/energy in the gas, although the concentrations required by gas turbines today are even smaller, and therefore, solid and gaseous impurities must be removed (SCHMITZ, 2000).

3. Experiments with biofuels in gas turbine engines

This item will be described some experiences with the utilization of biofuels in gas turbines.

Testing gas micro-turbines operating with biodiesel were performed by LOPP *et. al.* (1995); MIMURA (2003); BIST (2004); SCHMELLEKAMP and DIELMANN (2004); WENDIG (2004) and CORRÊA (2006). LOPP (1995) presented the results of the thermal performance of a gas turbine using a blend of jet fuel (Jet fuel - JF) and soy biodiesel (B). Three different fuels: JF, B10/JF90 and B20/JF80. The turbine efficiency reached its nominal value with B20, with a slightly better performance than with B10. There was an increase in fuel consumption proportional to the addition of biodiesel in the blend. The blends with biodiesel/diesel fuel were compatible enough to allow additional testing and show its potential as an alternative fuel. CO_2 emissions were reduced after the engine was fueled with B20. The two blends B10 and B20 did not show a noticeable increase in emissions of particulate matter compared to the JF. Subsequent inspections in the combustor and turbine blades showed no deposition or degradation of components.

MIMURA (2003) conducted a study of performance and emissions from a micro-turbine supplied with biodiesel from waste food oil regenerated. Operating in cogeneration mode the system had a thermal efficiency of 64%. It was observed emissions of 6 ppm of CO, 23 ppm of NO_X and 1.0 ppm of SO_X.

BIST (2004) performed a feasibility study on the use of methyl esters (biodiesel) derived from soy oil as additives blended with fuel gas turbine aircraft. Several blends were tested to identify which would meet the specifications for this type of engine without the need to change the initial design. Was not noticed a significant increase in fuel consumption for blends of B2, B5 and B10. In cases of B20 and B30 blends the increase in consumption was evident, being 7 and 10%, respectively. It was shown that a decrease in the efficiency of combustion in gas turbine as the percentage of biodiesel in the blend increase. In terms of emissions, an increase of CO content in the gases, due to the increase in the percentage of biodiesel in the blend, was noticed what indicates a reduction in combustion efficiency. This behavior is contrary to that seen in piston engine in which the CO decrease with increasing content of biodiesel blends.

There was also an increase in emissions of NO by increasing the concentration of biodiesel, therefore, NO_2 emissions did not change significantly. The high viscosity of biodiesel is a limiting factor for not testing with larger percentages than B30.

The biodiesel from soybeans contain glycerin. The amount of glycerin in the mixture should be kept as low as possible to avoid problems in combustion process. These measurements indicated that the B30 blend showed a content of glycerin which can be considered negligible. In none of the blends tested has been observed increases in pressure drop through analysis in the fuel filter and therefore none of them produced sediments that could cause blockage in the supply system of the turbine engine (BIST, 2004).

SCHMELLEKAMP and DIELMANN (2004) present the results of using vegetable oil from rape seed in a 30 kW micro-turbine, in blends of 10, 20 and 30%. Fuel consumption increased with increasing biodiesel blend. Using B30 obtained a 12% higher consumption in the range of operation. By using B10 it was verified that CO were lower than in the case of use of fossil diesel. However, when were employed mixtures of B20 and B30 it was observed a higher level of CO emissions. The viscosity of vegetable oil is much greater than that of biodiesel, so it is necessary to make a preheating the blend prior to injection.

The results of operating a 75 kW micro-turbine, burning biodiesel from rapeseed, sunflower, animal fats, were presented by WENDIG (2004). The operation with these three kinds of methyl esters showed a significant increase in emissions of CO and CO_2 at full load. All fuels examined showed a reduction in NO_X emissions in the range of 55%. Were problems related to the corrosive characteristics of biodiesel.

More recently, tests of performance and emissions in a 30 kW micro-turbine, using biodiesel from castor beans, were published by CORRÊA (2006). During the tests it was necessary to preheat the blends to 40 °C to achieve the viscosity values required by the manufacturer. The specific fuel consumption increased nearly 21% when using B100. Throughout the power range studied, it was observed a reduction in emissions of CO and NO_X in the exhaust gases.

WENDIG (2004) by using biogas found a decrease in CO with increasing load, reaching 100 ppm at full load. Since the emission of CO_2 was about 45 ppm, constant throughout the range of operation, and SO_2 emissions decreased. At full load the SO_2 emission was practically zero. The micro-turbine showed the lowest emission of NO_X at full load (20 ppm).

The existing reports on the use of biodiesel in micro-turbines describe increases in fuel flow rate when increasing the proportion of biodiesel in the blend; this is explained in part by lower LHV of biodiesel compared to diesel.

Problems due to viscosity, corrosion and accumulation of foreign material in the turbine blades may be in extended operations. These problems can be mitigated through a rigorous verification of the characteristics of biodiesel, demanding that it complies with the standards and manufacturers recommendations. In Brazil, ABNT published standards NBR 15341, NBR 15342, NBR 15343 and NBR 15344, to specify the properties of biodiesel (ABNT, 2006).

Manufacturers of gas turbines and micro-turbines are currently developing models that can use biofuels such as biodiesel. The changes that are taking place mainly consist in the use of materials resistant to corrosion in fuel and injection systems, and develop systems adjusted to work with injection of fuel physical-chemical characteristics different from diesel.

4. Cycles with potential for use of biofuels

A gas turbine is a set of three components: the compressor, combustion chamber and the turbine itself. This configuration forms a gas thermodynamic cycle accordance with the model ideal Brayton cycle is called. This set can operate in an open cycle, with air as the working fluid, which is admitted to the atmospheric pressure, passes through the turbine and is discharged back into the atmosphere without returning to the admission.

Thus, despite being an open cycle, some energy from the combustion is rejected in the form of heat contained in hot exhaust gases. The heat rejection is a physical limit of the gas turbine, intrinsic to the operation of thermodynamic cycles, even in ideal cases.

The loss of condition ideal cycle in a gas turbine can be quantified by the ratio involving the calorific value of fuel, discounting the power to drive the compressor and power net. Thus, decreasing the losses as it reduces the exhaust temperature, and raises the temperature of turbine inlet. The resistance to high temperature components of gas turbine is a very critical point in building technology such equipment (COHEN, H. *et. al.*, 1996).

Turbines designed to operate in simple cycle, as shown in Figure 1, in view of the thermal efficiency of the cycle, have gas outlet temperature reduced to maximum and have optimized compression ratio. The compression ratio is the ratio between the pressure of air entering and exiting the compressor. For example, if air enters at 1.0 atm, and leaves the compressor at 15.0 atm, the compression ratio is 15:1.

Apart from variation of the simple cycle obtained by the addition of these other components, considerations must be given to two system distinguished by the use of open (Figure1) and close cycles (Figure 2).

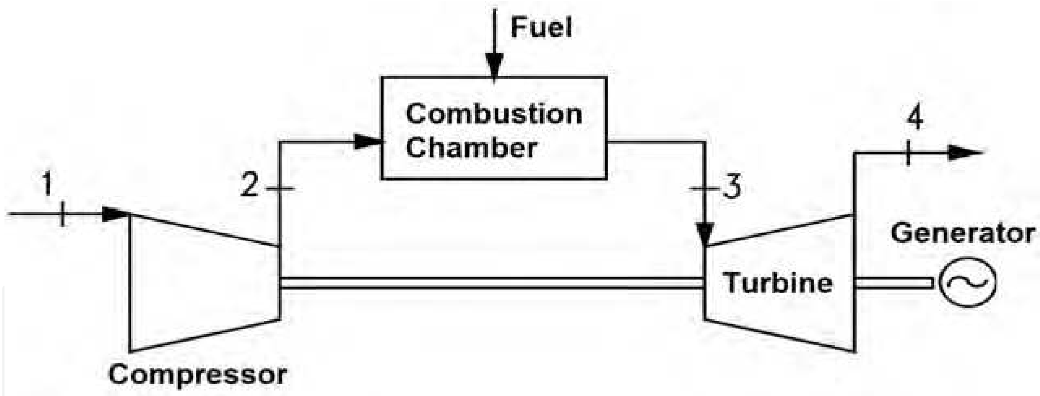

Fig. 1. Scheme of a simple cycle gas turbine system.

In the process of closed cycle operation is the same as the open cycle, the difference is that the working fluid remains within the system and the fuel is burned outside the system.
The biggest advantage of the closed cycle is the possibility of using high pressure throughout the circuit, which results in reducing the size of the turbomachinery, depending on power output, and allows the variation of power output by varying the pressure level of the circuit.
The significant feature is that the hot gases produced in the boiler furnace or reactor core never reach the turbine; they are merely used indirectly to produce an intermediate fluid, namely steam.
In order to produce an expansion through a turbine a pressure ratio must be provided and the first necessary step in the cycle of gas turbine plant must therefore be compression of the working fluid.

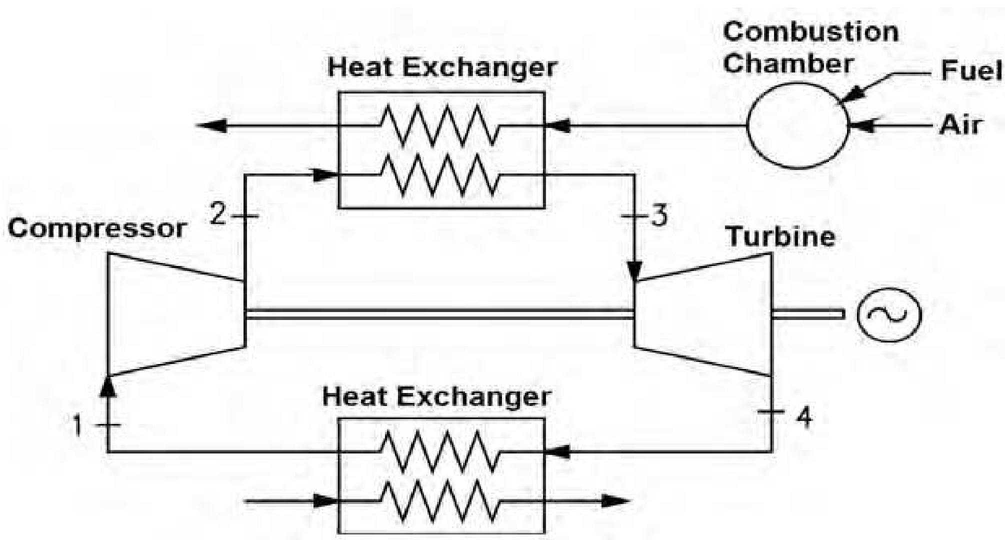

Fig. 2. Scheme of a close cycle gas turbine with a heat exchanger.

If after compression the working fluid was to be expanded directly in the turbine, and there were no losses in either component, the power developed by the turbine would just equal that absorbed by the compressor. Thus if the two were coupled together the combination would do no more than turn itself round. But the power developed by the turbine can be increased by the addition of energy to raise the temperature of the working fluid prior to

expansion. When the working fluid is air a very suitable means of doing this is by combustion of fuel in the air which has been compressed.
One way to increase the thermal efficiency of a closed cycle is the addition of a heat exchanger; however there are limits to the introduction of this equipment because it can cause loss of pressure in the circuit.
A cycle gas turbine with a heat exchanger can also be called a regenerative cycle, as shown in Figure 3, wherein the heat rejected in the exhaust gases of the gas turbine passes through the heat exchanger and heats the air from the compressor before entering the chamber combustion. The pre-heated air reduces the fuel consumption injected into the combustion chamber, increasing the thermal efficiency of the cycle.

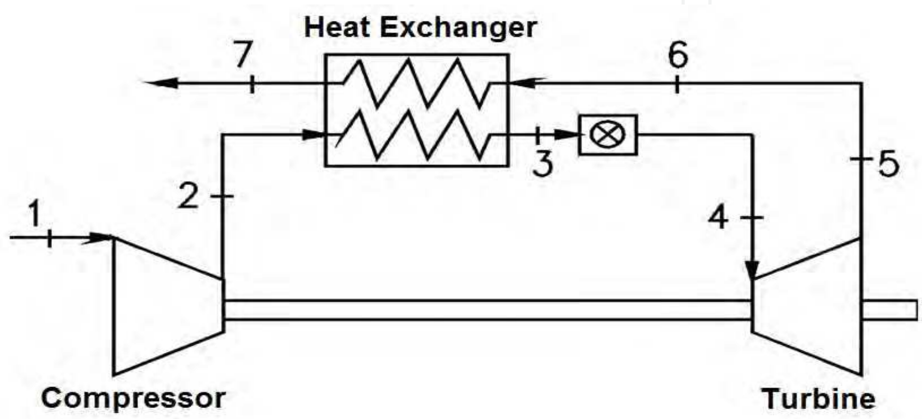

Fig. 3. Scheme of a regenerative cycle gas turbine.

Specific gas turbines to operate in combined cycle, as shown in Figure 4, are developed in order to maximize the thermal efficiency of the cycle as a throughout. Therefore, reducing the temperature of the exhaust gas is not necessarily the most critical point in terms of efficiency, since the gas turbine exit are still used to generate power in other equipment.

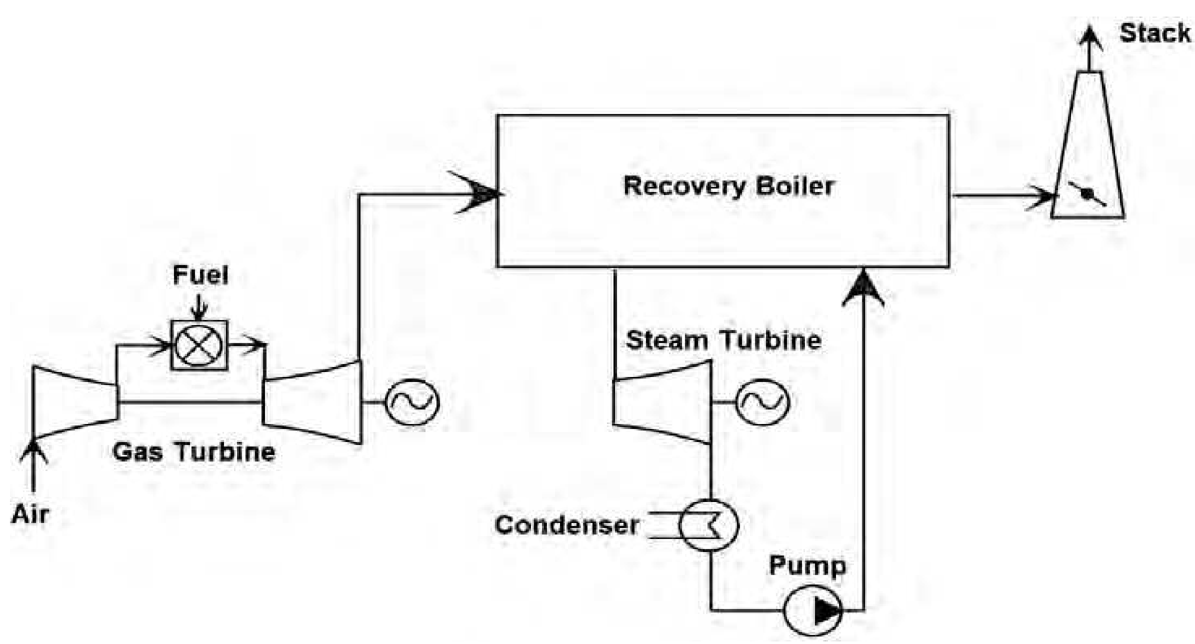

Fig. 4. Scheme of a combined cycle.

Another configuration of installation of gas turbines is the cycle where there is the presence of an intercooler before or between compressors, i.e., compressors among the low and high pressure. In this configuration the air that enters into the first compressor is compressed to a pressure intermediate between the maximum of the cycle and the ambient pressure, as shown in Figure 5.

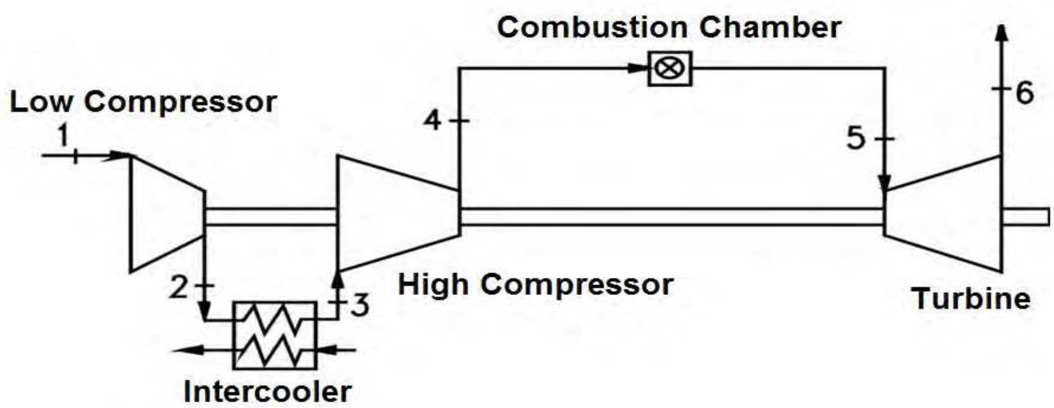

Fig. 5. Scheme of a cycle with intercooler.

In the case of plants with two compressors the air leaving the first compressor (low pressure) enters the intercooler where the heat is removed without major pressure drop. Being that, in practice, the air is returned to the circuit should have a temperature slightly higher than the entrance to the first compressor.
In this type of gas turbine is possible to operate with temperatures above 1450 °C due to the reduction of air temperature of cooling the turbine blades, increasing the thermal efficiency of the cycle, reaching 50%.
The combined cycle consists of one or more gas turbine, whose exhaust gases are injected into a recovery boiler that provides steam to a turbine.
In an open cycle, the thermal efficiency is low, around 30 to 35%. In a combined cycle efficiency can reach 60% (the highest efficiency of all types of driver). In combined cycle occurs the combination of gas turbines with steam turbines.

4.1 Selection cycle gas turbine during operation with gas biofuel

In simple cycle the thermal efficiencies is increased with the increase of pressure ratio and turbine inlet temperature. Therefore, high pressure ratios require high pressure to clean the gas, which results in high costs of equipment, and high compression energy consumption.
In a cycle of regeneration is evident with a peak thermal efficiency at a pressure ratio between 4 and 8. In relation to biomass gasification, the cycle with regenerator is interesting because it is possible to use moderate pressure gasification resulting in lower equipment costs and acceptable thermal efficiencies compared to the simple cycle.
Preheating inlet air aerator does not increase much thermal efficiency, since the output of the reactor; higher temperatures are derived fuel gas. However, this represents a contradiction, since the positive effects of preheating are reversed, it is necessary to perform the cooling of the gas for cleaning.
With gas pyrolysis cycle gas turbine with a regenerator would lower overall efficiency, because the residual heat from the pyrolysis process cannot be used to increase the power of the turbine.

The combined cycle integrated with slow pyrolysis uses a Rankine cycle supplied with charcoal and a cycle gas turbine powered by pyrolysis gas, which results in a high cycle efficiency, because a considerable amount of waste heat can be used to produce Rankine cycle power (SCHMITZ, 2000).

4.2 Adjustments to gas turbine

Due to impurities in the gas or fuel, for example, the synthesis or biofuel, it is necessary redesign the gas turbine combustor. For each type of fuel, run a kind of optimization, with reference to a low value of the LHV of fuel.
To compensate for the lower value of LHV for the fuel gases, the fuel injection system must provide a fuel rate much higher than when the combustor operates fuel with high calorific value. Due to the high rate of mass flow of gas with low LHV, the passage of fuel has a much larger cross section than the section corresponding to natural gas. The fuel pipes and control valves and stop valve have larger diameters and shall be designed to include an additional fuel blend, which consists of the final mixture of the recovered gas with natural gas and steam. The pressure drop and the size of the spiral of air entering the flame tube were adjusted to optimize the combustion process. The system must have high safety standards, so the flanges and gaskets of the combustor and its connections must be good soldiers. The system for low fuel LHV must include:

- Fuel line with a low LHV;
- Natural gas line;
- Steam line to reduce NO_X;
- Line blending of fuel with low LHV;
- Line of nitrogen to purge;
- Lines pilot;
- Compressor;
- Combustion Chamber.

The loading of the gas turbine to the rated load is accomplished through the use of the fuel reserve for security reasons. The procedure for replacing the fuel reserve for the main runs automatically. The characteristics of blends are monitored and analyzed online.

5. Case study with biodiesel and ethanol in gas turbine

For perform the tests in the gas turbine was built a test bench in the laboratory of gas turbines and gasification of the Institute of Mechanical Engineering, Federal University of Itajubá - IEM/UNIFEI. This bench is made of a micro-turbine Capstone C30 model cycle with regenerative power of 30 kW, configured to operate with liquid fuel.
This gas turbine is used mainly for primer power generation or emergency and can work in a variety of liquid fuels. This turbine uses a recovery cycle to improve its efficiency during operation, due to a relatively low pressure what facilitates the use of a single shaft radial compression and expansion (BOLSZO, 2009).
The experimental tests were carried out by using a 30 kW regenerative cycle diesel single shaft gas turbine engine with annular combustion chamber and radial turbomachineries, whose characteristics at ISO conditions are given in Table 3 (CAPSTONE, 2001).
To perform the experiments with different fuel blends and it was implemented a system to preheat the fuel supply aiming to control the viscosity of the fuel used.

For tracking and measuring the parameters of the test was used a supervisory software in test bench (given by the manufacturer of the turbine) and a data acquisition and post processing of data obtained during the tests.

Fuel Pressure	350 kPa
Power Output	29 kW NET (± 1)
Thermal Efficiency	26% (± 2)
Fuel HHV	45,144 kJ/kg
Fuel Flow	12 l/h
Heat Rate (LHV)	14,000 kJ/kWh
Exhaust Temperature	260 °C
Inlet Air Flow	16 Nm^3/min
Rotational Speed	96000 rpm
Pressure Ratio	4

Table 3. Engine Performance data at ISO Condition.

5.1 Tests on gas turbine using biodiesel fuels

To perform the experimental tests were used blends biodiesel/diesel, including the total replacement of diesel with biodiesel in the gas turbine. The blends considered in the experiment were: B10, B20, B30, B50 and B100. Due to low solubility in diesel fuel at low temperature tests with ethanol were performed without pre-mix, and also without the use of additives, which enhance the cost of fuel.

Following the methodology of the measures to be adopted to test gas turbine (item 2), Table 4 shows the physical-chemical properties of biodiesel for testing thermal performance and emissions:

Properties	Soy Biodiesel	Palm Oil Biodiesel	Diesel	Micro Turbine Manufacturer Fuel Limits	ASTM D6751
Cetane Number	58	60	45.8	-	> 47
Sulfur (% mass)	0	0	0.20	0.05 <	< 0.05
Kinematic Viscosity @ 40 °C (mm^2/s)	2.19	2.26	1.54	1.9 – 4.1	1.9 – 6
Density @ 25 °C (g/cm^3)	0.888	0.854	0.838	0.75 – 0.95	-
Flash Point (°C)	136	138	60	38 - 66	> 130
Water (% Volume)	0.05	0.05	0.05	0.05	0.05

Table 4. Biodiesel and diesel physic-chemical characteristics.

Table 4 shows the comparison amongst the characteristics of commonly used types of biodiesel and diesel pure, with the requirements of the fuel made by the manufacturer of gas turbine tested and specifications for biodiesel fuel blend of standard ASTM D6751.

Once obtained the blends, there was the experimental determination of the calorific value, kinematic viscosity and density of different blends (B10, B20, B30, B50, B100) and ethanol, according to the standards ISO 1928-1976 and ASTM D1989-91, respectively. In the case of viscosity were made ten measurements to reduce the standard deviation, and the calorific value were performed five measurements for each blend. Table 5 shows the LHV of the fuels used in the experiment.

Fuel	Pure (kJ/kg)	B10 (kJ/kg)	B20 (kJ/kg)	B30 (kJ/kg)	B50 (kJ/kg)
Biodiesel (Palm Oil)	37,230.37	41,204.29	40,691.27	39,768.17	38,623.27
Biodiesel (Soy)	28,298.16	41,425.27	40,748.82	39,864.36	39,061.45
Diesel	42,179.27	-	-	-	-
Alcohol	23,985.00	-	-	-	-

Table 5. Lower Heating Value of fuels used.

The use of different fuels implies the need to make adjustments of mass flow rate of them, according to its LHV and your density, because without these adjustments, once established a load, the supply system would feed a quantity of fuel depending on the characteristics of standard fuel (diesel). If the LHV of the new fuel is lower than the standard, the gas turbine power could not reach the demanded.

Initially, the engine was operating with conventional diesel fuel for a period of 20 minutes to reach a steady state condition for a load of 10 kW. After 20 minutes, the mass flow rates were changed to the corresponding values of blends diesel/biodiesel. At this stage, it begins to replace the fuel, in order of increasing content of biodiesel (B10, B20, B30, B50 and B100), closing the inlet valve of diesel and opening the valve of the mixture. In order to ensure that all existing diesel power on the internal circuitry of the engine would be consumed, the engine was left operation for 10 minutes with the same load operation (10 kW).

In order to check if the fuels were able to feed the engine, without experiencing any problems, regarding the fuel injection system, the kinematic viscosity of each fuel was measured. The composition of gas emissions and thermal parameters were also measured in total and average load for each fuel. This whole procedure was performed for the engine operating with loads of 5, 10, 15, 20, 25 and 30 kW in a grid connection mode.

Afterwards it was held on measurement of emissions with gas analyzer, and increased the load of 5 kW, 10 minutes waiting again until it reaches steady state again.

When finished testing with a blend, the engine was left running, in order to accomplish the purging of fuel remaining. After executing the purge the supply system it was loaded with a new mixture. Once completed the tests with biodiesel and blends, was returning to operate the engine with diesel for ten minutes, and then it was disconnected and stopped.

5.2 Thermal performance

The results of performance testing of a 30 kW gas turbine engine supplied with biodiesel from palm oil, soy and ethanol are shown:

Figures 6, 7 and 8 show the relationship between specific fuel consumption and power. The graphs correspond to soy biodiesel, Figure 6, biodiesel from palm oil, Figure 7 and diesel and ethanol, Figure 8.

In the case of biodiesel from soy, Figure 6 observed if an increase in specific fuel consumption, when increasing the fraction of biodiesel in the blend in the range of 10 to 25 kW. The lowest value occurs when using pure diesel oil, the highest value occurs when using pure biodiesel and the difference between the curves of diesel and B100 remains approximately constant at 12.82%. This behavior if repeated when using biodiesel from palm oil as shown in Figure 7. When using ethanol, Figure 8 also shows an increase in specific fuel consumption with respect to diesel.

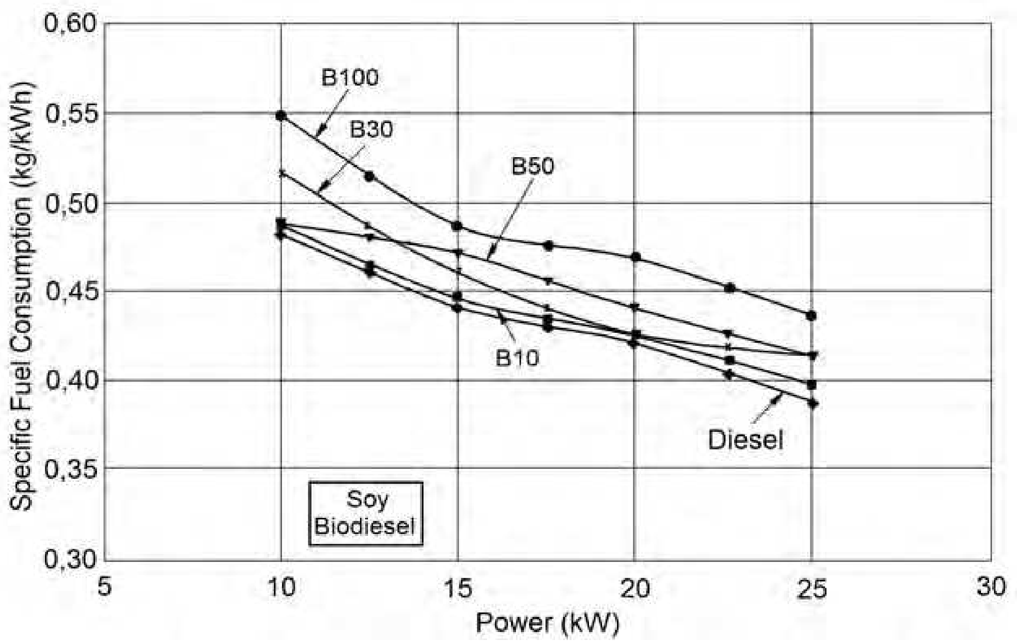

Fig. 6. Specific fuel consumption versus power of the gas turbine for different fuels: soy biodiesel.

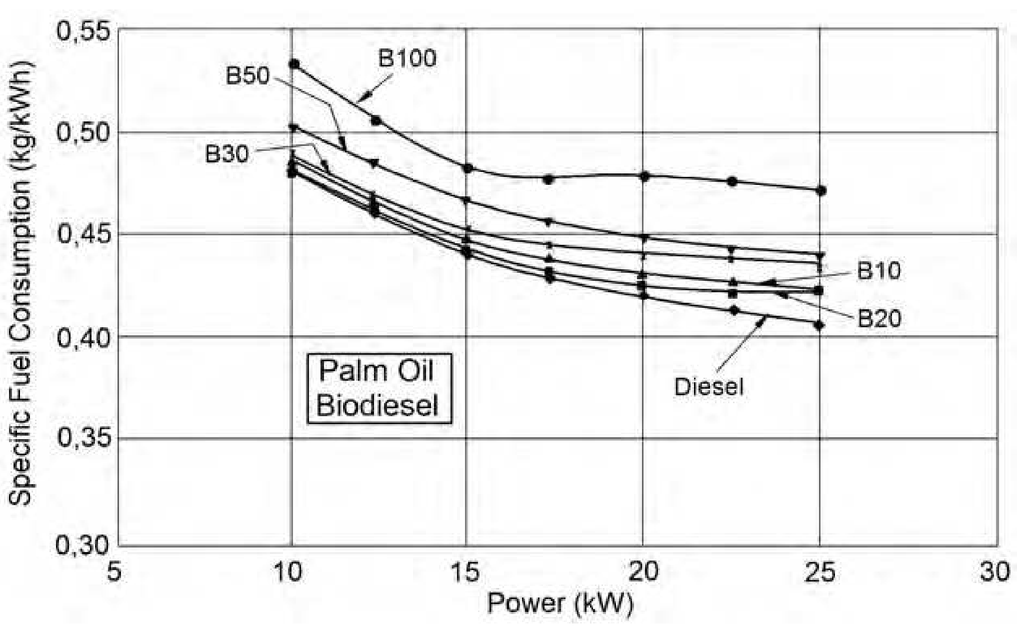

Fig. 7. Specific fuel consumption versus power of the gas turbine for different fuels: palm oil biodiesel.

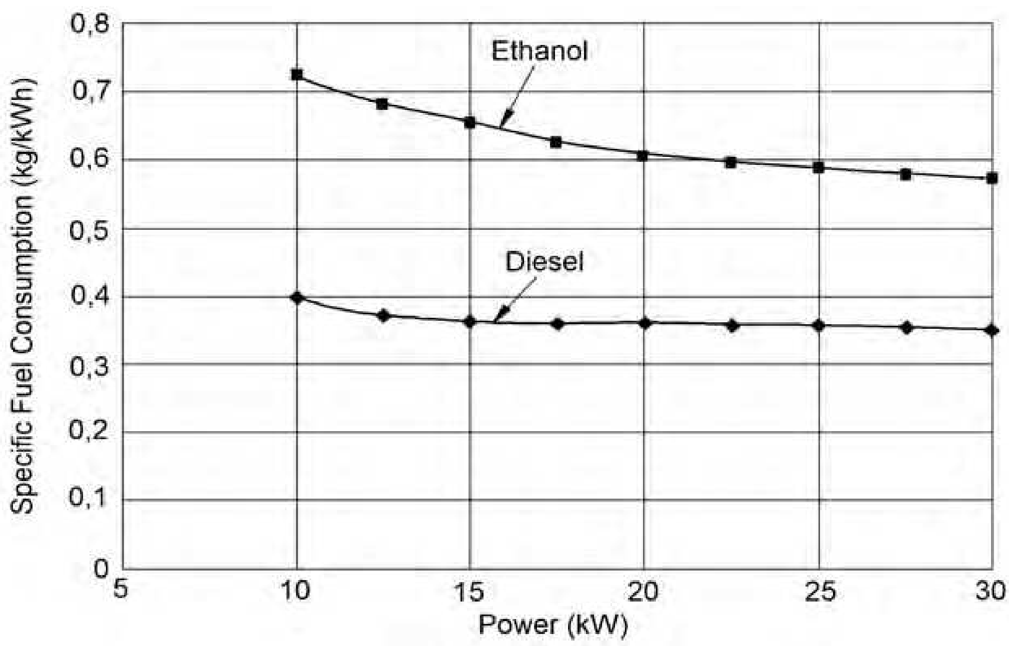

Fig. 8. Specific fuel consumption versus power of the gas turbine for different fuels: diesel and ethanol.

The specific consumption of biodiesel from soy and palm oil were approximately equal. Consumption already of ethanol was higher due to even lower calorific value of fuel compared with the others used fuel.
Figures 9, 10 and 11 show the behavior of power versus Heat Rate of gas turbine for the three different fuels tested.

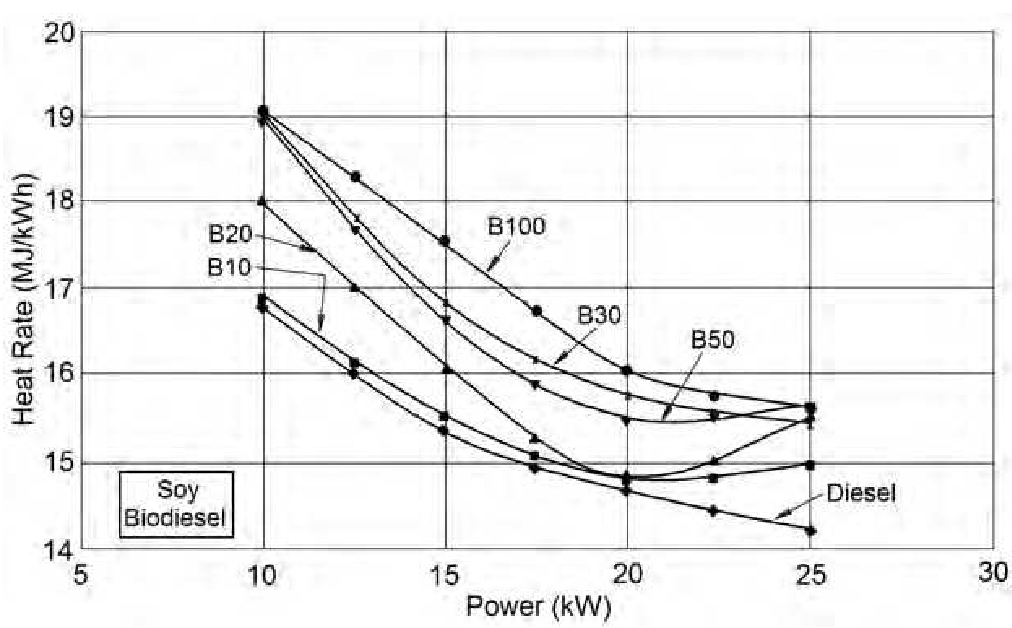

Fig. 9. Heat Rate versus power for different fuels: soy biodiesel.

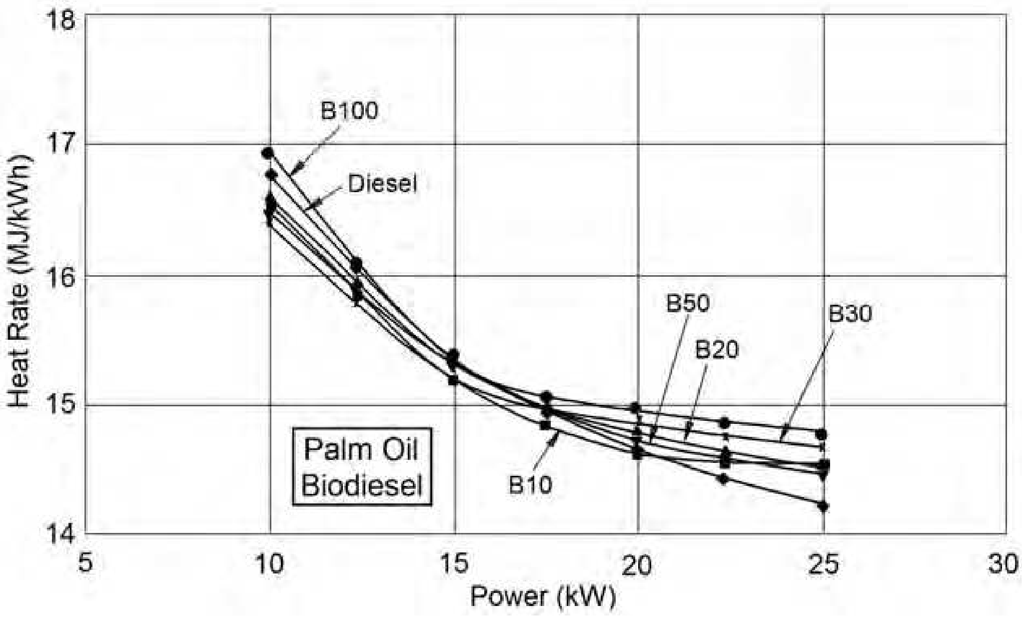

Fig. 10. Heat Rate versus power for different fuels: palm oil from biodiesel.

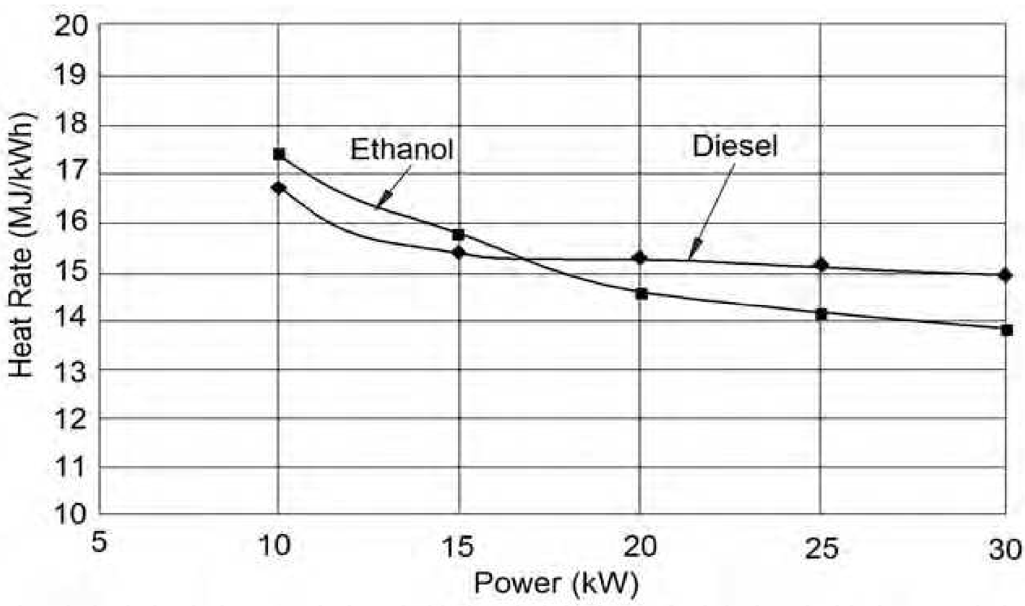

Fig. 11. Heat Rate versus power for different fuels: diesel and ethanol.

In all cases, it was observed a reduction in Heat Rate with the increase in power. Operation with biodiesel from palm oil presented a better performance than soy biodiesel (lower value of the Heat Rate). Similarly to the specific fuel consumption, the differences in the amount of Heat Rate between diesel and B100 remained approximately constant in the range 10 to 25 kW. The lower fuel consumption occurs to the rated power when operating with diesel. This is due to the higher calorific value of diesel.

When a lower LHV of fuel is used, a greater mass of fuel is needed to release in the combustion energy required for a specific power. The mass flow rate of fuel passing through turbine increases and the compressor operating point changes, making their efficiency and, consequently, the cycle efficiency decreases.

Finally, Figures 12, 13 and 14 displays the graphs of the thermal efficiency of the gas turbine for different fuel and power operation. The differences between the efficiency with diesel and biodiesel blends with soy biodiesel, Figure 7 may be due to differences in density, viscosity and LHV of fuel compared with conventional biodiesel. This can cause changes in the process of atomization of the fuel within the combustion chamber, reducing the thermal efficiency of the engine as previously mentioned.
In the case of palm oil biodiesel was not observed differences, and the efficiency values remain equal to those obtained when the engine was tested with diesel, within the entire power range evaluated. Ethanol showed the lowest efficiency among the fuels tested

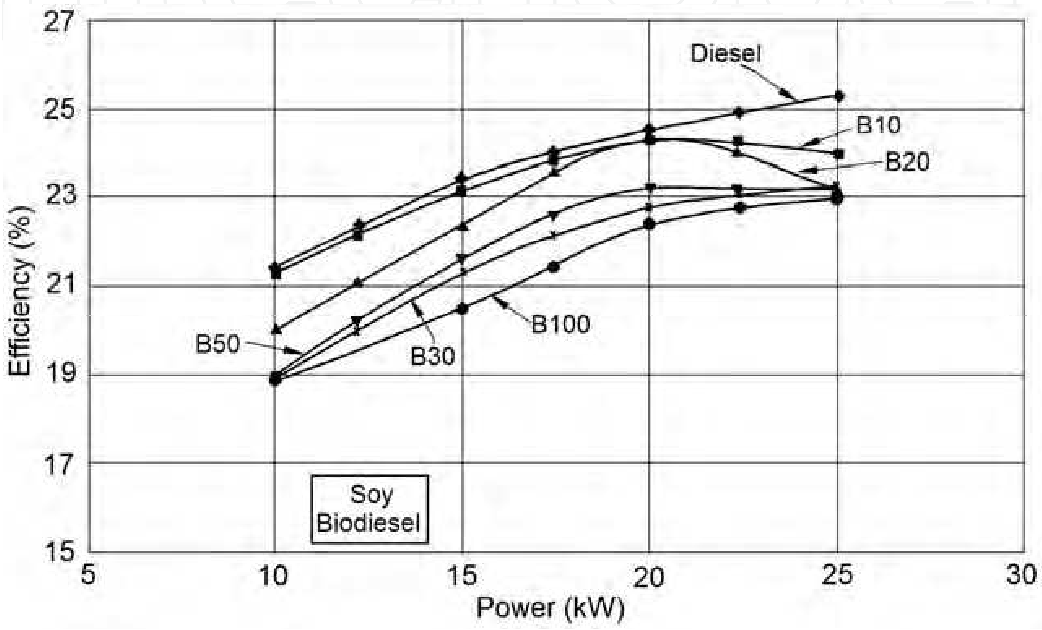

Fig. 12. Efficiency versus power turbine for different fuels: soy biodiesel.

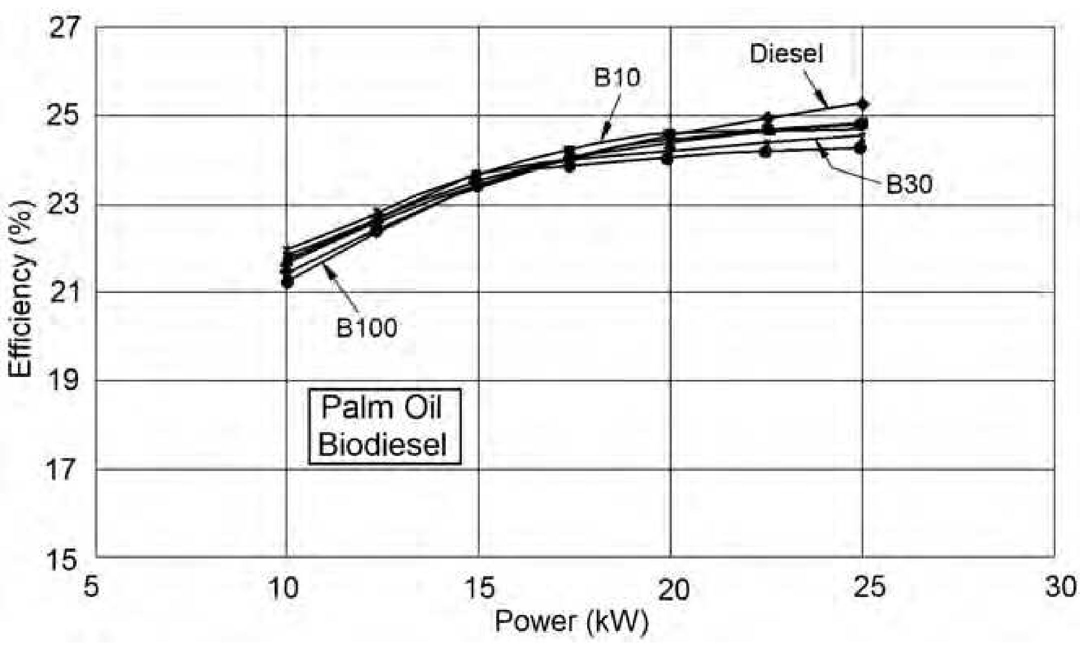

Fig. 13. Efficiency versus power turbine for different fuels: palm oil biodiesel.

As in tests performed by HABIB (2010), the test conducted in the laboratory of UNIFEI with B100 biodiesel resulted in high thermal efficiency compared to other blends, as shown by the graphs of Figures 12 and 13, such performance is attributed to equivalence ratio, which produced the best ratio of air to fuel during firing, resulting in more complete combustion due to the presence of extra oxygen in the biofuel, which resulted in the presence of 16 to 19% oxygen in the gases turbine exhaust.

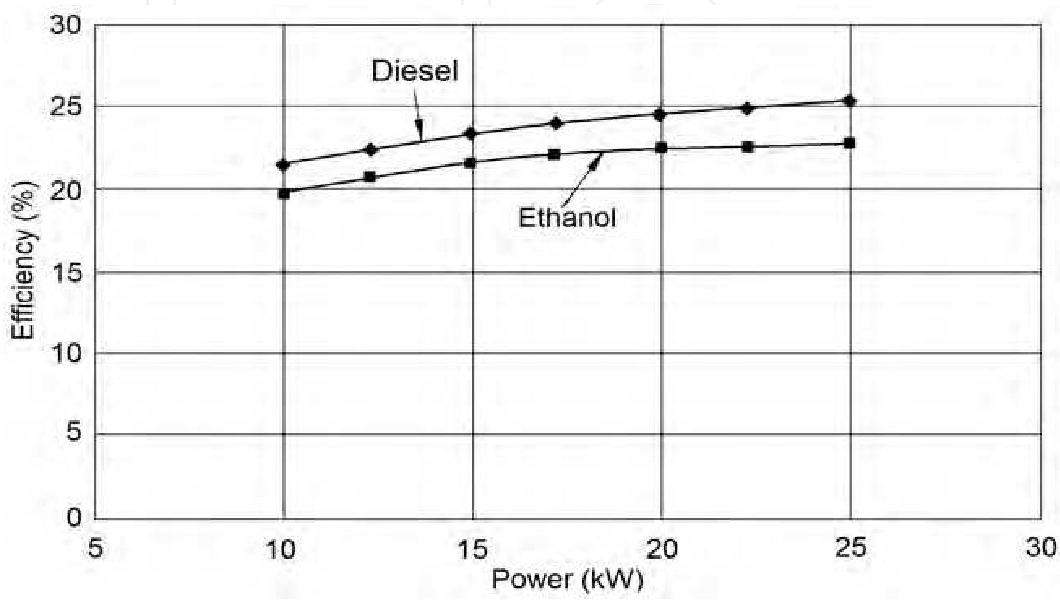

Fig. 14. Efficiency versus power turbine for different fuels: diesel and ethanol.

As mentioned by other authors, it can say that the type of biodiesel is an important factor in the analysis of projects for power generation, as small differences in density, viscosity and LHV cause changes in the parameters of thermal performance of turbines gas, and amongst them when compared with pure diesel.

For the case of palm oil biodiesel, although there is an increase in specific fuel consumption, there were no significant differences in the efficiency of the engine. With soy biodiesel there were differences in the efficiency of around 2% throughout the power range tested.

The results obtained with ethanol were very different for both types of biodiesel tested, due to its lower calorific value.

In practical terms, for distributed generation, ethanol must go through economic and technical assessments in order to detect.

5.3 Emission of pollutants

Publications on experiments with biofuels in gas turbines are not yet sufficient to make definitive conclusions in terms of emissions. However, it is observed that the CO decreases with increasing load. The opposite happens with the NO_X emissions. It is also provided a reduction in the emission of smoke, along with the increase in the emission of NO_X.

The emission results shown in the following are results of experiments with three different biofuels: Soy biodiesel, palm oil biodiesel and ethanol achieved in the laboratory of UNIFEI. The first two were blended with diesel in different proportions and each mixture was tested with a gas micro-turbine operating at full load and partial loads. The pure ethanol was used and compared with the performance of pure diesel, as reported in the results performance.

The analysis focuses mainly on changes in the levels of carbon monoxide (CO) and nitrogen oxides (NO_X), unburned hydrocarbons were not detected in the combustion products, during the tests. This is due to high temperatures and high excess air into the combustion chamber. Likewise there were no emissions of sulfur oxides (SO_X), since biofuels evaluated did not contain sulfur in your composition.

Figures 15, 16 and 17 shows the concentrations of CO in the exhaust gas from the micro-turbine gas to biodiesel blends: soy biodiesel, Figure 15, palm oil biodiesel, Figure 16 and pure diesel and ethanol, Figure 17.

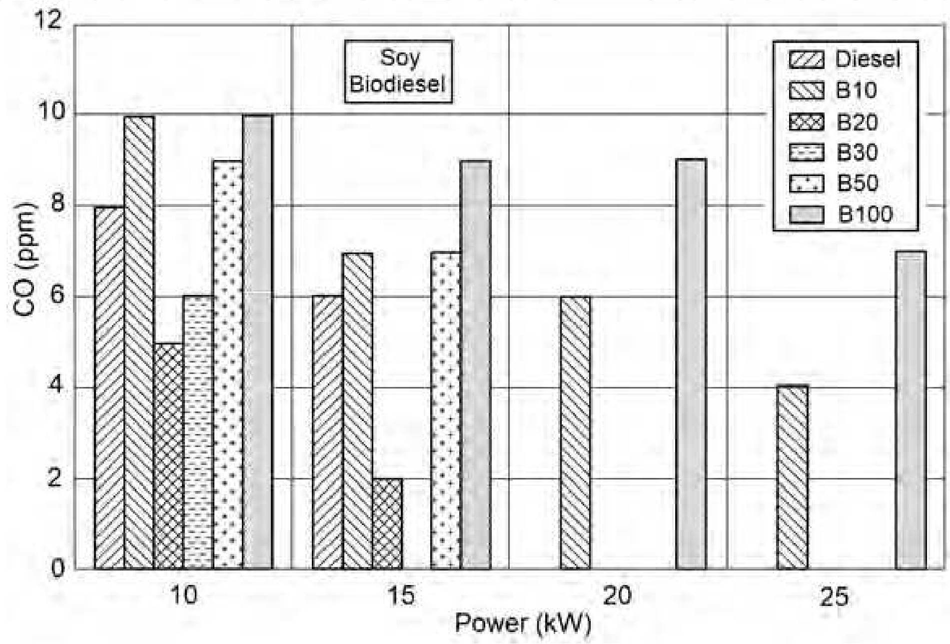

Fig. 15. Concentrations of CO in gas versus the micro-turbine power for different fuels: soy biodiesel.

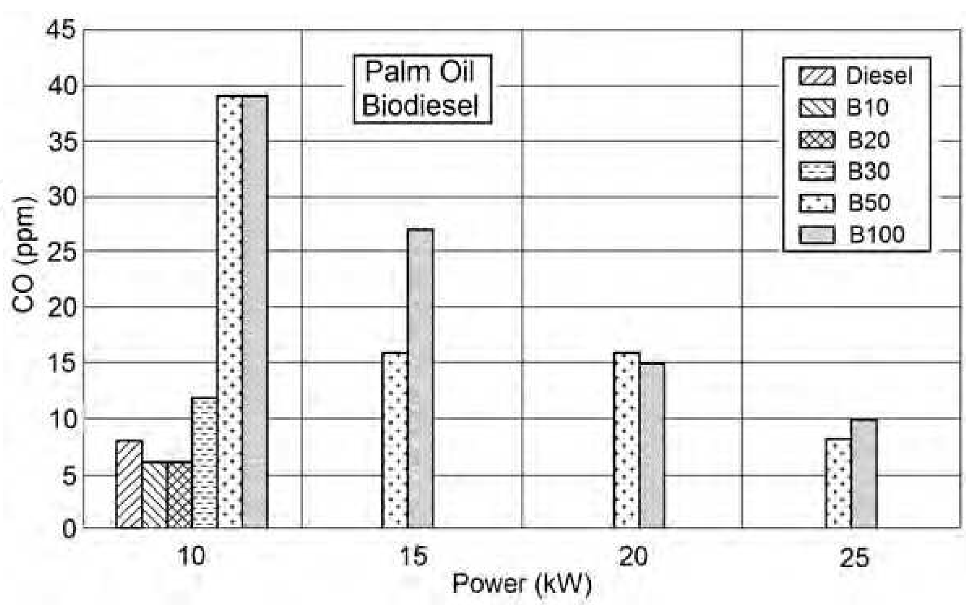

Fig. 16. Concentrations of CO in gas versus the micro-turbine power for different fuels: palm oil biodiesel.

It is observed in Figures 15, 16 and 17 that the concentrations of CO in the exhaust of the micro-turbine operating on pure biodiesel (B100) are higher than when operating with diesel. Emissions decrease as much for diesel as for the blends when the power increases. This if explained by the characteristics of combustion in the combustion chamber.

The soy and palm oil biodiesel has a higher viscosity than the pure diesel. The fuel nozzles not modified in the micro-turbine, must have worsened the quality of atomization of biodiesel compared with diesel, generating higher levels of CO in the exhaust gases as consequence of incomplete combustion. CO emissions at part load are larger than at full load. The lower emission occurs to full load.

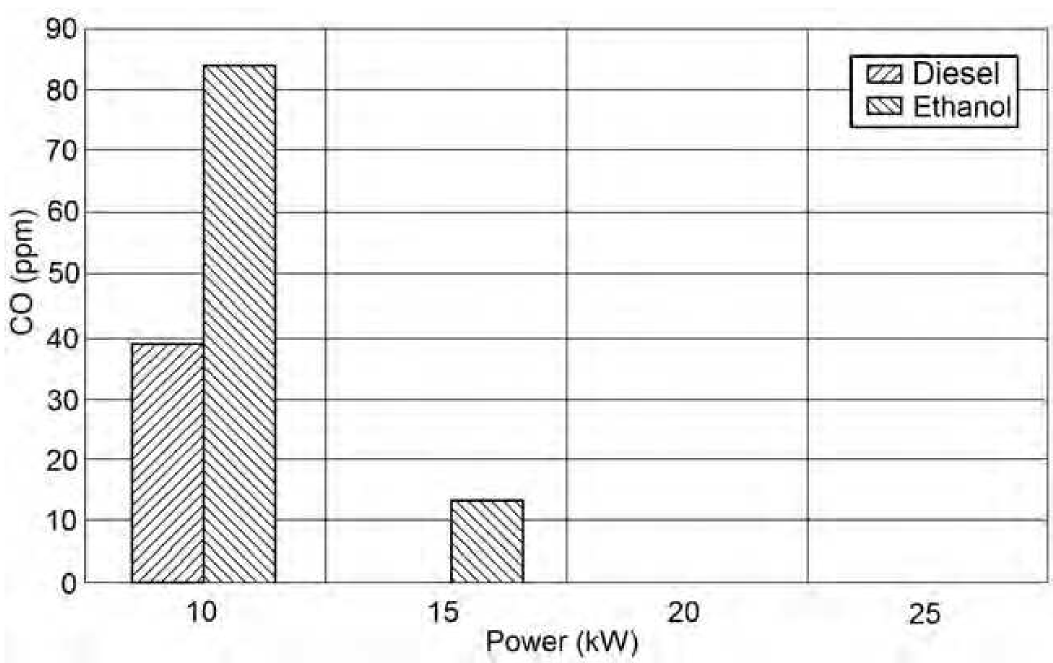

Fig. 17. Concentrations of CO in gas versus the micro-turbine power for different fuels: pure diesel and ethanol.

There are differences in the composition of exhaust gases by using soy and palm oil biodiesel, as can be seen in Figures 15 and 16, as well as been observed previously in thermal efficiency. The operation with soy biodiesel showed no CO in loads exceeding 10 kW, while the operation with palm oil biodiesel (B100) presented CO throughout the power range. The two fuels have different physical-chemical characteristics, which are reflected in a particular behavior in the combustor process.

It is observed in Figure 17 that the emissions of CO to ethanol use are higher than diesel, in all power. This is probably due to lower LHV of ethanol relative to diesel, resulting in higher specific fuel consumption, which reduces the residence time of fuel in the combustion chamber, which may be the cause of greater incomplete combustion.

Figures 18, 19 and 20 show NO_X emissions with different fuels. There are no great differences in the values of NO_X for the fuels.

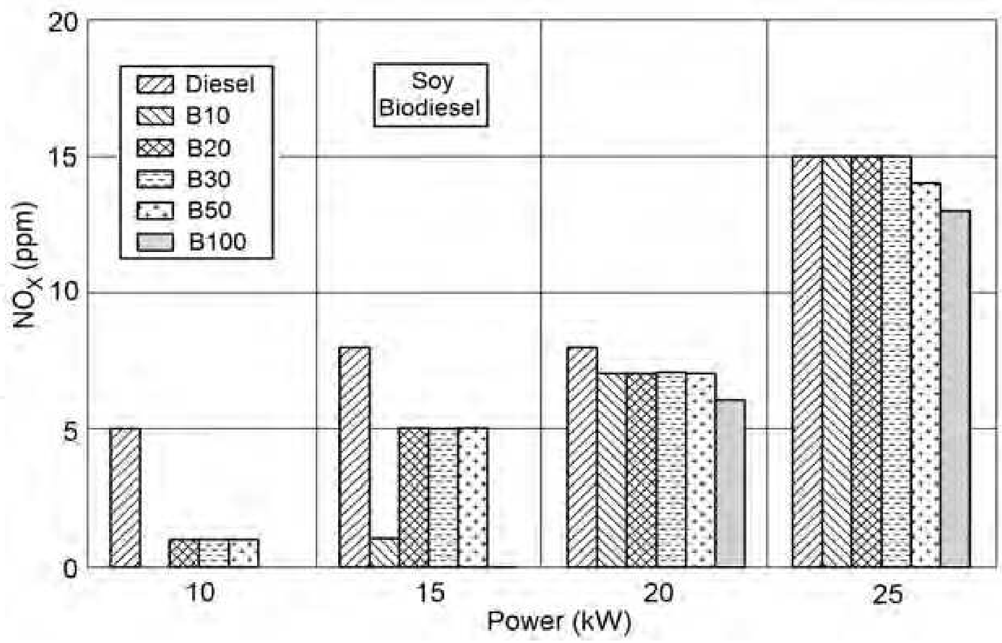

Fig. 18. NO_X emissions versus the micro-turbine power for different fuels: soy biodiesel.

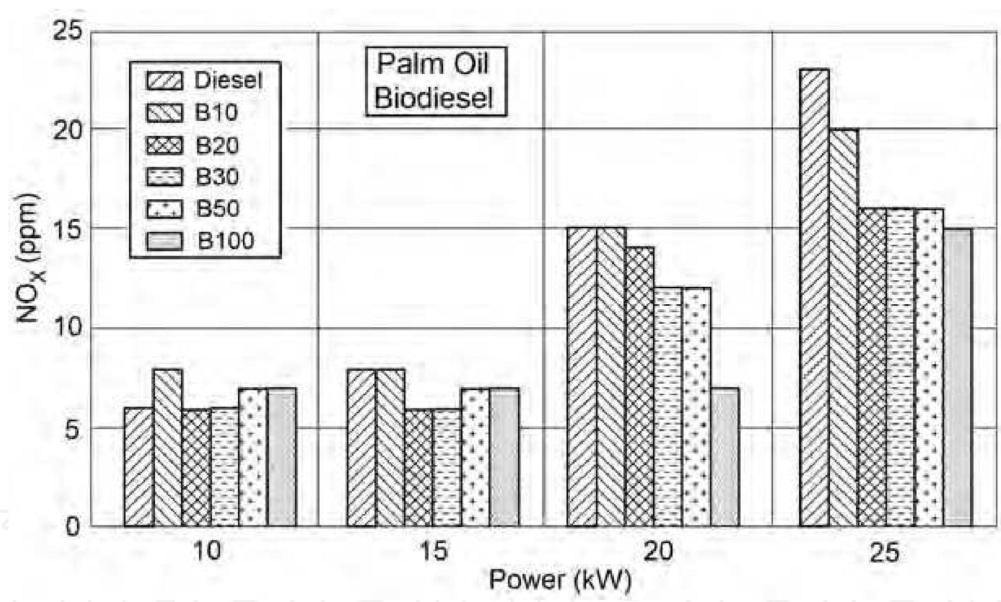

Fig. 19. NO_X emissions versus the micro-turbine power for different fuels: palm oil biodiesel.

NO_X emissions are predominantly of thermal origin and their values are in all cases below 35 ppm, maximum limits set by the engine manufacturer. When used pure biodiesel (B100), the NO_X concentration is lower than the diesel at all loads tested. The results for CO and NO_X show a behavior similar to that presented by PETROV (1999), who also has carried out experiments with a 30 kW micro-turbine.

In the case of ethanol, NO_X emissions showed an inverse behavior to CO, which was expected, because the formation of CO and NO_X is a function of reaction temperature when the CO reduces NO_X increases. Thus, when the amount of CO decreases with increasing power, increases the quantity of NO_X in exhaust gases.

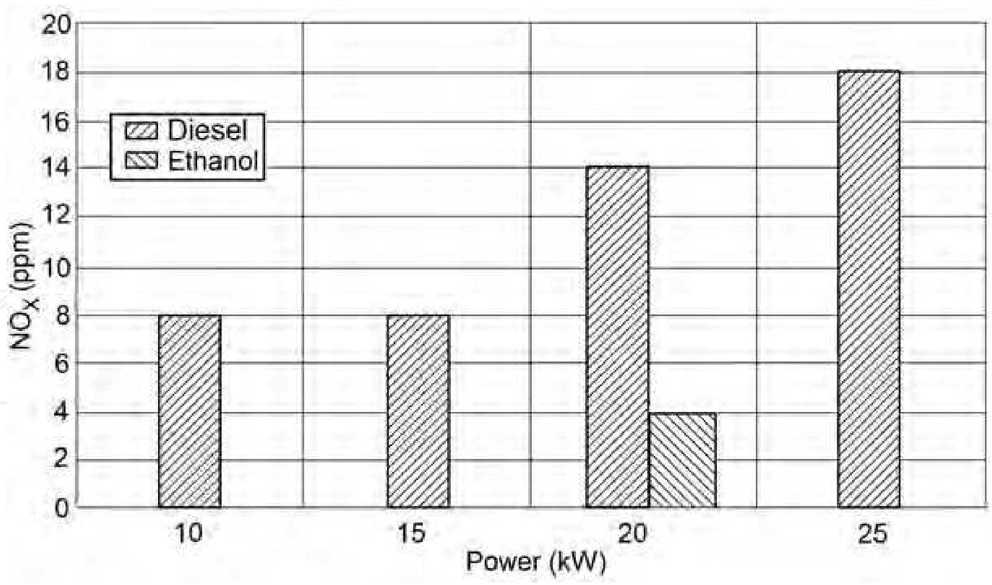

Fig. 20. NO_X emissions versus the micro-turbine power for different fuels: pure diesel and ethanol.

For the three fuels evaluated it was observed a reduction in emissions of NO_X and an increase in CO content compared with diesel, however, all results are within the range indicated by the manufacturer (CAPSTONE, 2001).
It is remarkable that there were no SO_2 emissions and unburned hydrocarbons in any test with biofuels, as mentioned.

6. Conclusions

With increasing industrial development there is the necessity for more refined research on the use of biofuels in gas turbines, covering aspects such as quality of biofuel to be used, form and storage conditions, adjustments in the systems of engines, costs of energy generated in order to maintain high operational reliability of the turbines.
A gas turbine can operate with different types or blends of biofuels with a corrected power loss at around 4.26%, and the corrected heat rate of 8.38% higher than diesel fuel, as shown in this work.
As previously warned care must be taken during the operation of the gas turbine with liquid fuel (from whatever source) and gaseous fuels derived from biomass, because the components of supply systems of gas turbines are very sensitive and the use these biofuels can provide wear or cause loss of efficiency during extended operation.
The physicochemical characteristics of all the fuels evaluated lie within the specifications for their use in gas turbines. The thermal performance tests showed that biodiesel has higher specific consumption than diesel. The reason for that is the lower heat value of the pure biodiesel in comparison with diesel fuels. Agreeing with the findings of other researchers, cited in this work, and verified in tests with biofuels the fuel that presented the smallest difference in terms of heat rate in relation to diesel was the palm oil biodiesel, with a difference of about 17.6 % at full load and less than 1.0 % at medium load.
The tests also found the need to take care with the installations of storage systems and supply of power plants in order to maintain the specifications and properties of mixtures of liquid biofuels within acceptable standards. In the case of gaseous fuels to be careful with

the filter system of gas particles and elimination of harmful substances such as tar and others that, besides causing wear to components of the supply system, undermine the performance of engines.

Emission levels from the experimental tests have shown that CO increases for the palm oil biodiesel and NO_X decreases by approximately 26.6%; SO_X concentration wasn't taken into account when biodiesel was used. Further investigation involving emission has to be carried out for the better understanding of pollutant formation when biodiesel fuels are used.

7. Acknowledgements

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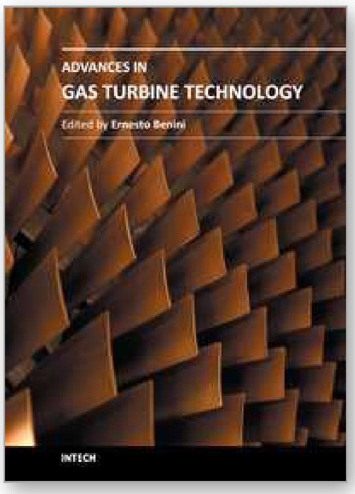

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